Experimental Investigation on Wall Friction Factor in a Vertical Annulus Channel under Natural Circulation Water Flow Conditions

Seongbae Park [a*], Youngchang Ko [a], Byongjo Yun [a]
[a]*Nuclear Systems Division, Mechanical Engineering Dept, Pusan National Univ., busan daehak ro 63, geumjeong gu, Busan*
[*]*Corresponding author: tjdqo963@pusan.ac.kr*

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1. Introduction

Recently, the adoption of passive safety systems based on the Natural Circulation(NC) flow is increasing to enhance the safety of the nuclear power plants. These passive systems are essential to remove core decay heat in the accident of failure of reactor coolant pump. The NC flow is governed by the buoyancy of coolant resulted by wall heat transfer and the pressure drop including wall friction factor. However, the prediction of pressure drop using 1-D system codes relies on wall friction factor models developed for Forced Circulation(FC) conditions, such as the Colebrook[1] equation. Those models do not fully account for the characteristics of NC flow and create challenges in prediction. In the present study, an experiment was conducted to investigate the wall friction factor in vertical annulus under single-phase NC flow conditions.

2. Experimental Set-up

2.1 Experimental Facility

The experimental apparatus, shown in Fig.1, is a closed-loop system that includes a vertical annulus test section(heater), heat exchangers(cooler), pump, preheater and accumulator. This loop is designed to operate with water temperatures and pressures up to 235℃ and 30 bars, respectively.

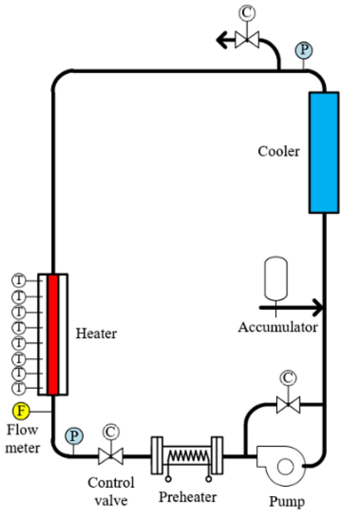

Fig. 1. Schematic diagram of experimental loop

Fig.2 illustrates the test section with an inner diameter of 15.3 mm and having a heating rod with an outer diameter of 9.5mm in the channel center. Along the axial direction, K-type thermocouples of which diameter is 0.5 mm are installed with 100 mm interval to measure fluid temperature. The length of the heated section is 804 mm. Pressure drop is measured by using a Differential Pressure(DP) transmitter over a 260 mm section at the end of the test section. The pressure impulse lines connected to the DP transmitter have a 30 cm horizontal section to cool down the fluid. Additionally, a fiber optical sensor is installed at the top of the test section to detect whether boiling occurs or not during experiments.

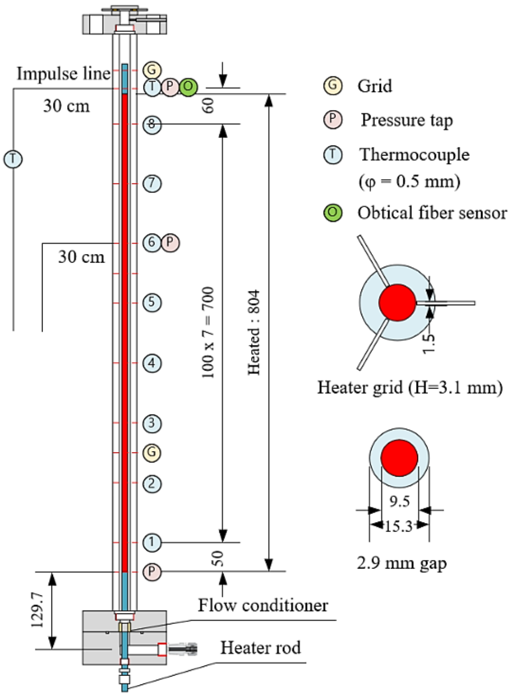

Fig. 2. Schematics of the test section

The total height of the loop is 4.7 m and difference of elevation between the centers of the heating and cooling sections is 2.4 m. A preheater of which maximum power is 50 kW is equipped to establish the preset inlet coolant temperature. An accumulator, using nitrogen gas, regulates the pressure of the loop with a pressure control accuracy of 0.1 bar.

2.2 Data Reduction Method

The differential pressure along the test section is expressed in the steady-state conditions as follows.

$$(1)\ \Delta P = \left(f\frac{L}{D_h}+K\right)\frac{\rho_{avg}v_{avg}^2}{2} + \left(\rho_{avg}-\rho_{I,avg}\right)gL$$

Where ρ_{avg} and $\rho_{I,avg}$ represent the average densities of the working fluid in the test section and the impulse line, respectively. In the above equation, the first term on the right side indicates the permanent pressure loss in the test section. The second term corresponds to the hydrostatic head difference between the test section and pressure impulse line.

3. Experimental results

3.1 Adiabatic Forced Circulation Experiment

Forced circulation experiments were conducted to verify the experimental setup and measurement system. The experiments were performed at 24℃ of inlet fluid temperature under atmospheric pressure condition. Fig. 3 shows the comparison of the experimental data and existing models and it confirms that present measurement system is reliable.

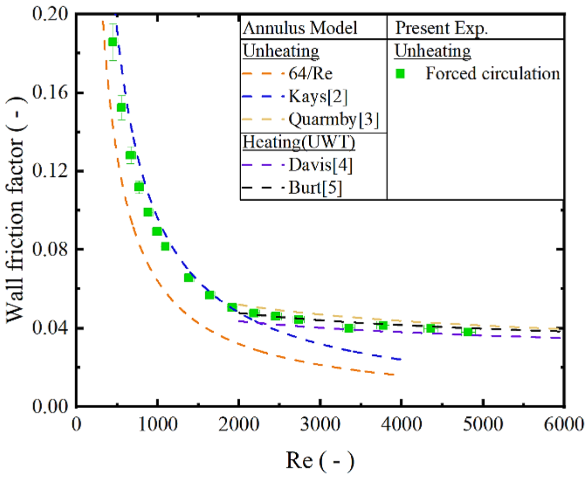

Fig. 3. Adiabatic forced circulation experimental results

3.2 Natural Circulation Experiment

Natural circulation experiments were performed at the condition of inlet temperature ranging from 27 to 33℃, pressure of 6.5 bar and heat flux ranging from 19 to 557 kW/m^2.

In Fig.4, the experimental results are compared with the existing models applicable to forced circulation conditions. The experimental data shows a decreasing trend according to Reynolds number. Fluctuation of measured wall friction factors is observed in the low Re condition. It is stabilized and begin to diminish at a Reynolds number of 1150 and converged at a certain value beyond a Reynolds number of 1500. The convergence of friction factors implies the transition from laminar to turbulent flow. The flow regime transition is confirmed clearly by the fluid temperature distributions measured at Reynolds number of 1450. As shown in the Fig. 5, fluid temperature fluctuations exceeding 15℃ are observed.

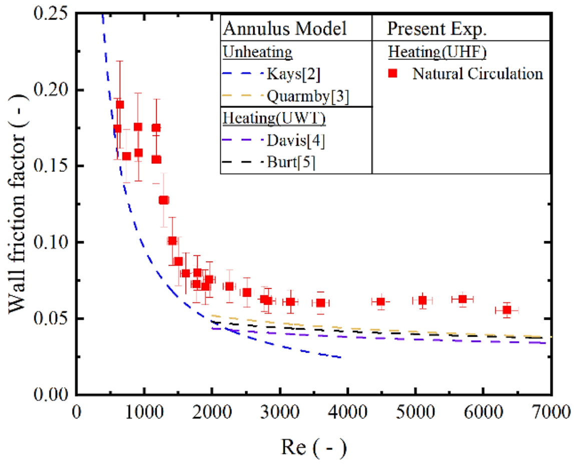

Fig. 4. Natural circulation experimental results

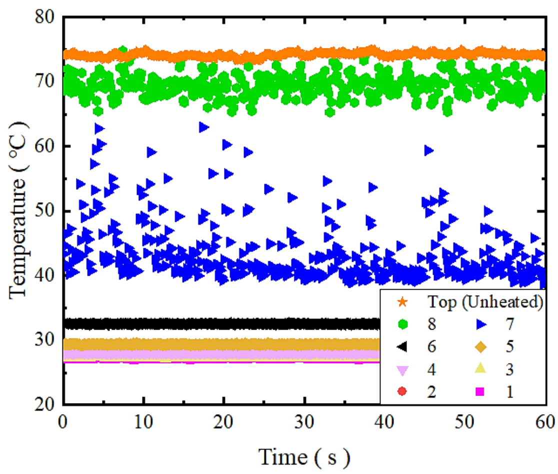

Fig. 5. Fluid temperature distribution at Reynolds number of 1450)

In Fig.4, The measured wall friction is higher than that of existing models. The experimental data was generated using uniform heat flux conditions and the models are developed using the experimental data of adiabatic and Uniform Wall Temperature(UWT) conditions. The different heating conditions affect the thermal properties and flow characteristics near the wall. According to Ndenguma et al.[6], the friction factor in an annular channel under mixed convection decreased in the order of cooling > heating > isothermal heating conditions. Additionally, Tsou et al.[7] presented that the friction factor rises as a Grashof number increases at a given Reynolds number. These results demonstrate the effect of wall temperature and buoyancy on friction factor. Therefore, the difference between present data and the models is mainly caused by the different heating condition which affects the thermal properties and flow characteristics.

4. Conclusions

Wall friction factors were investigated natural circulation experiments in a vertical annulus channel. From the experiment, it was observed that the wall friction factor significantly decreased as Re increases until Re = 1,500, and then decreased gradually as Re increased further. Finally, it reached to the asymptotic value at Re of 3,000.

Present experimental data was used as benchmark data for the evaluation of the existing models. It reveals that the existing models underestimate the present data. It also showed well that the necessity of development of a new wall friction model considering viscosity and buoyancy near the wall for an accurate prediction of natural circulation flow.

NOMENCLATURE

$\Delta \boldsymbol{P}$	Differential pressure [Pa]
$\boldsymbol{f}$	Friction factor [-]
$\boldsymbol{L}$	Measurement length [m]
$\boldsymbol{D_h}$	Hydraulic diameter [m]
$\boldsymbol{K}$	Form loss factor [m]
$\boldsymbol{\rho}$	Density [$\boldsymbol{kg/m^3}$]
$\boldsymbol{v}$	Velocity [$\boldsymbol{m/s}$]
$\boldsymbol{T}$	Temperature [℃]
$\boldsymbol{Re}$	Reynolds number [$\boldsymbol{\rho v D_h/\mu}$]
$\boldsymbol{\mu}$	Viscosity [Pa·s]

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